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Fang Liu
Purdue University

Eckhard A. Groll
Purdue University

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Analysis of a Two Phase Flow Ejector for the Transcritical CO₂ Cycle

Fang LIU^{1*}, Eckhard A. GROLL²

Purdue University, School of Mechanical Engineering,
West Lafayette, IN, USA

¹Tel: 765-494-0231, Fax: 765-494-0787, E-mail: liu39@purdue.edu

²Tel: 765-496-2201, Fax: 765-494-0787, E-mail: groll@purdue.edu

*Corresponding Author

ABSTRACT

This paper presents the analysis of a two phase flow ejector for the transcritical CO₂ cycle. A detailed simulation model of a two phase flow ejector was developed. A controllable ejector expansion device was designed, constructed, and installed in a transcritical CO₂ experimental air conditioning system. System-level experimental results were obtained at different operation conditions and various ejector geometries. The ejector expansion model was then utilized to determine the efficiencies of the motive nozzle, suction nozzle, and mixing section using the system-level measured data. It was found that motive nozzle efficiency decreases as ejector throat area decreases and that the suction nozzle efficiency is affected by the outdoor temperature and ejector throat area. In addition, the distance from the motive nozzle exit to the mixing section constant area entry not only affects the suction nozzle efficiency, but also affects the mixing section efficiency.

1. INTRODUCTION

CO₂ is being advocated as one of the natural refrigerants to replace CFCs and HCFCs in vapor compression systems due to its environmentally friendly characteristics. However, the lower efficiency of the basic transcritical CO₂ refrigeration cycle compared to vapor compression systems using HFC and HCFC refrigerants is a major hindrance for the technology to make progress towards practical applications. In order to recover the expansion losses and increase the cycle efficiency, it has been proposed to replace the expansion valve with an ejector expansion device. The ejector expansion device offers the advantages of simplicity, reliability and availability compared to work producing expansion devices. Literature studies have shown that the COP of the ejector expansion transcritical CO₂ cycle can be improved by 7 to 22% over the basic transcritical CO₂ cycle for typical air conditioning operation conditions using assumed ejector efficiencies or ejector components efficiencies of 0.7 to 0.9 (Jeong et al., 2004; Li and Groll, 2005; Ksayer and Clodic, 2006; Deng et al., 2007; Elbel and Hrnjak, 2008). In addition, it was found that the COP of the ejector cycle is very sensitive to the ejector efficiency (Domanski, 1995). However, the knowledge of the efficiency of two-phase flow ejectors is limited. In most of the literature studies, values of 0.7 to 0.9 were assumed for the individual ejector component efficiencies (Domanski, 1995; Alexis and Rogdakis, 2003; Elbel and Hrnjak, 2004; Yapici and Ersoy, 2005; Li and Groll, 2005; Ksayer and Clodic, 2006; Yu and Li, 2007; Deng et al., 2007; Elbel and Hrnjak, 2008). No studies were found in the literatures in which measured values of the ejector nozzle or mixing section efficiencies were obtained. Chaiwongsa and Wongwises (2007) studied experimentally the effect of throat diameters of the ejector on the performance of the refrigeration cycle using a two phase flow ejector as an expansion device. They found that in an R-134a system, a motive nozzle having a throat diameter of 0.8 mm yielded the highest COP, while a motive nozzle having a throat diameter of 1.0 mm yielded the lowest COP. However, literature studies about the effect of CO₂ ejector geometries on the ejector-expansion transcritical CO₂ cycle are also very limited. To obtain the ejector efficiencies for different geometries at different operation conditions and study the effects of the ejector geometries on the ejector cycle performance, a two phase flow ejector simulation model that takes the geometries into account has to be developed. The development of such a model and the calculation of the ejector efficiencies using this model are detailed in this paper.

2. MODELING

For single-phase flow ejectors, there are well established models to conduct performance analysis and design calculations (Keenan et al. 1950; Munday and Bagster 1977; Huang et al., 1999; Sun 1995). However, for two-phase flow ejectors, there are no established models to perform an analysis or design a device because of the complexity of the two-phase flow. A schematic of a two-phase flow ejector is shown in Figure 1. High pressure motive stream expands in the motive nozzle and entrains low pressure suction stream into the mixing section, which is treated in the same way as the expansion process of a converging nozzle to simplify the analysis. The two streams mix in the mixing section and become one stream then this stream increases its pressure in the diffuser. The simulation model of a two-phase flow ejector developed here combines submodels of the motive nozzle flow, suction nozzle flow, mixing section flow and diffuser flow.

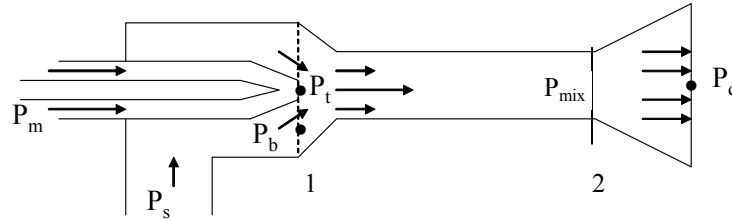


Figure 1: Schematic of the ejector working processes

2.1 Critical Flow Model of Two-Phase Flow

In the motive nozzle of the ejector, carbon dioxide initially at supercritical pressure and temperature expands into the sub-critical two-phase region. At the nozzle throat, the flow will become critical for typical operation conditions of an ejector expansion transcritical refrigeration cycle. A critical flow model of the two-phase flow must be established firstly to predict the performance of the motive nozzle. The critical flow model introduced here is established by applying Katto's principle for two-phase critical flow to one-dimensional one-component homogeneous equilibrium two-phase pipe flow (Katto 1968, Katto 1969). The expression for the speed of sound can be obtained as (Attou and Seynhaeve 1999):

$$V_c = \left(\frac{v_{mix}^2 (h_g - h_f)}{(v_g - v_f)(h'_{mix} - v_{mix}) - v_{mix}'(h_g - h_f)} \right)^{\frac{1}{2}} \quad (1)$$

It can be seen that the speed of sound given in Equation (1) depends only on pressure and quality.

2.2. Model of Motive Nozzle Flow

Based on the critical flow calculation introduced in Equation (1), a model for the motive nozzle of the ejector can be setup using the following assumptions:

- The flow inside the motive nozzle is a steady, one dimensional flow.
- The nozzle is a converging nozzle and its throat is at its exit.
- At the nozzle throat, the flow reaches the critical flow condition.
- The isentropic efficiency of the nozzle, η_m , is given
- The inlet flow velocity is neglected.
- The heat transfer between the fluid and nozzle wall is neglected.
- The gravitational force effect on the flow is neglected.

Considering that the fluid enters the motive nozzle at a pressure p_m and temperature T_m , the following model will predict the pressure p_t and velocity V_t at nozzle exit, which is also its throat. The isentropic efficiency of the nozzle is defined by:

$$\eta_m = \frac{h_m - h_t}{h_m - h_{t,is}} \quad (2)$$

where h_m is the enthalpy of inlet flow, h_t is the enthalpy of exit flow and $h_{t,is}$ is the enthalpy assuming an isentropic expansion from p_m to p_t . By assuming a value for the exit pressure p_t , $h_{t,is}$ can be determined from the inlet entropy s_i and pressure p_t . Thus, the enthalpy h_t can be calculated for a given nozzle efficiency η_m . The energy conservation

between the inlet and exit of the motive nozzle can be expressed as shown in Equation (3) in order to calculate the exit velocity, V_t :

$$h_t = h_i + \frac{V_t^2}{2} \quad (3)$$

For the assumed exit pressure p_t and the calculated h_t , the quality x_t can be determined. The speed of sound V_c can then be calculated based on Equation (1). In the next step, the speed of sound, V_c , is compared to the exit velocity, V_t , and the pressure p_t is updated until the iteration provides reasonable agreement. For a given throat area A_t , the mass flow rate through the motive nozzle can be determined by:

$$\dot{m}_m = \rho_t A_t V_t \quad (4)$$

where the flow density at the nozzle throat is calculated as follows:

$$\rho_t = \frac{1}{\frac{x_t}{\rho_{g,t}} + \frac{1-x_t}{\rho_{f,t}}} \quad (5)$$

In summary, the exit pressure and velocity of the motive nozzle is determined by the given inlet flow conditions and its isentropic efficiency. Using a specified throat area, the mass flow rate is determined as well.

When the mass flow rate through the motive nozzle is less than the critical mass flow rate as determined with the above model, the motive nozzle is operated under non-critical mode. With the mass flow rate as a given parameter, Equations (2) to (5) can be used to determine the exit pressure and velocity.

2.3. Model of Suction Nozzle Flow

In a real ejector, the suction nozzle is typically replaced by a suction chamber. However, to simplify the analysis, the expansion process from the suction inlet to the mixing section inlet is treated in the same way as the expansion process of a converging nozzle using the following assumptions:

- The flow is steady one-dimensional flow.
- The isentropic efficiency of the nozzle is given.
- The inlet flow velocity is neglected.
- The heat transfer between the fluid and nozzle wall is neglected.
- The gravitational force effect on the flow is neglected.

Once the mass flow rate through the motive nozzle has been determined, the mass flow rate through the suction nozzle can be determined using the ejection ratio φ as shown in Equation (6).

$$\dot{m}_s = \varphi \dot{m}_m \quad (6)$$

For a given inlet pressure p_s and enthalpy h_s of the suction nozzle, the exit pressure p_b and velocity V_b can be predicted for a specified isentropic efficiency η_s and an exit area A_b using the following procedure. The isentropic efficiency of the nozzle is defined as:

$$\eta_s = \frac{h_s - h_b}{h_s - h_{b,is}} \quad (7)$$

where h_s is the enthalpy of inlet flow, h_b is the enthalpy of exit flow and $h_{b,is}$ is the enthalpy for an isentropic expansion process from p_s to p_b . Assuming an exit pressure p_b , $h_{b,is}$ can be determined based on the inlet entropy s_s and exit pressure p_b . Using Equation (7), the actual exit enthalpy h_b can be calculated for a given isentropic efficiency η_s . The energy conservation equation between the inlet and exit of the suction nozzle can be expressed as:

$$h_s = h_b + \frac{V_b^2}{2} \quad (8)$$

With the assumed exit pressure p_b and the calculated exit enthalpy h_b , the density ρ_b can be determined. The exit velocity V_b can be calculated based on mass conservation equation:

$$\dot{m}_s = \rho_b A_b V_b \quad (9)$$

In the next step, the exit velocity V_b calculated from Equation (8) is compared to the exit velocity calculated from Equation (9) and the exit pressure p_b is updated until the iteration provides reasonable agreement. For typical operating conditions, the critical flow condition will not be reached in the suction nozzle because of the small pressure difference between p_s and p_b .

2.4. Model of Mixing Section Flow

The ejector mixing section starts from the exits of the motive nozzle and the suction nozzle to the exit of the mixing section as shown in Figure 1. To simplify the model of the mixing section, the following assumptions are made:

- At the inlet plane 1, the motive stream has a velocity of V_t , a pressure of p_t , and occupies the area A_t .
- At the inlet plane 1, the suction stream has a velocity of V_b , a pressure of p_b , and occupies the area A_b .
- At the outlet plane 2, the flow becomes uniform and has a velocity of V_{mix} and a pressure of p_{mix} .
- The motive stream pressure and suction stream pressure keep unchanged from the nozzle exits until the inlet of the constant area mixing section. There is no mixing between the motive stream and suction stream before the inlet of the constant area mixing section.
- The heat transfer between the fluid and the mixing section wall is neglected.
- The friction between the fluid and the mixing section wall is neglected.
- The gravitational force effect is neglected.

Using the above assumptions, the model to predict the mixing stream velocity V_{mix} and pressure p_{mix} based on the motive stream velocity V_t and pressure p_t , and the suction stream velocity V_b and pressure p_b can be established as follows. The mass conservation equation between the inlet plane and outlet plane reduces to:

$$\rho_t A_t V_t + \rho_b A_b V_b = \rho_{mix} A_{mix} V_{mix} \quad (10)$$

where ρ_{mix} is the density of the mixing stream at the outlet plane. The mixing section efficiency η_{mix} was used to account for the frictional losses of the mixing chamber (Huang et al., 1999, cited by Elias, 2007). With an assumed η_{mix} of the mixing section, the momentum conservation equation between the inlet plane and outlet plane reduces to:

$$p_t A_t + \eta_{mix} \rho_t A_t V_t^2 + p_b (A_{mix} - A_t) + \eta_{mix} \rho_b (A_{mix} - A_t) V_b^2 = p_{mix} A_{mix} + \rho_{mix} A_{mix} V_{mix}^2 \quad (11)$$

The energy conservation equation between the inlet plane and outlet plane reduces to:

$$\dot{m}_t \left(h_t + \frac{V_t^2}{2} \right) + \dot{m}_s \left(h_b + \frac{V_b^2}{2} \right) = (\dot{m}_t + \dot{m}_s) \left(h_{mix} + \frac{V_{mix}^2}{2} \right) \quad (12)$$

Based on the thermophysical property relationships of the fluid, the density ρ_{mix} can be determined from the pressure p_{mix} and enthalpy h_{mix} . Thus, the pressure p_{mix} , velocity V_{mix} and enthalpy h_{mix} can be calculated from Equations (10), (11) and (12).

At the exit plane of the mixing section, the fluid will be in the two-phase region for typical operating conditions of the ejector-expansion transcritical carbon dioxide cycle. The quality of the mixing stream can be determined from its pressure and enthalpy. The speed of sound of the two-phase mixing stream can then be calculated using Equation (1) to see if the critical flow condition is reached.

2.5. Model of Diffuser Flow

In the diffuser, the kinetic energy of the mixing stream will be converted to a static pressure increase. By assuming that the mixing stream at the outlet of the mixing section is a homogeneous equilibrium flow, a pressure recovery coefficient, C_t can be defined as:

$$C_t = \frac{p_d - p_{mix}}{\frac{1}{2} \rho_{mix} V_{mix}^2} \quad (13)$$

where p_d is the pressure at the exit of the diffuser. A correlation proposed by Owen et al. (1992) is used here to calculate the pressure recovery coefficient from the area ratio of the diffuser as follows:

$$C_t = 0.85 \rho_{mix} \left[1 - \left(\frac{A_{mix}}{A_d} \right)^2 \right] \left[\frac{x_{mix}^2}{\rho_{g,mix}} + \frac{(1 - x_{mix})^2}{\rho_{f,mix}} \right] \quad (14)$$

where x_{mix} is the quality of mixing stream at the diffuser inlet, and $\rho_{g,mix}$ and $\rho_{f,mix}$ are the saturated vapor and liquid densities at pressure p_{mix} , respectively. A_d is the exit area of the diffuser. By neglecting the heat loss from the ejector to the environment, the enthalpy at the diffuser outlet h_d can be determined from the energy conservation equation of the whole ejector as follows:

$$\dot{m}_t h_t + \dot{m}_s h_s = (\dot{m}_t + \dot{m}_s) h_d \quad (15)$$

The quality at the diffuser outlet x_d can then be determined from the exit pressure p_d and exit enthalpy h_d using the thermophysical property relationships of the fluid.

2.6. Discussion of Two-Phase Ejector Modeling Results

By combining the models of the motive nozzle flow, suction nozzle flow, mixing section flow and diffuser flow, a simulation model of a two-phase flow ejector has been developed. The model uses a specified motive nozzle throat area and efficiency, suction nozzle efficiency, cross sectional area of the mixing section, mixing section efficiency, and exit area of the diffuser. The model predicts the pressure, quality and mass flow rate of the outlet stream for given inlet conditions of the motive stream and suction stream, and a given ratio of the mass flow rates between these two streams. The ejector simulation model was used to investigate the effects of the design parameters of the ejector and the operation conditions on the performance of the ejector.

A higher diffuser exit pressure is desired in an ejector expansion refrigeration cycle as it means a higher compressor inlet pressure. A higher quality at the diffuser exit means that less liquid refrigerant enters the evaporator and that leads to a smaller refrigeration capacity. Therefore, a low quality and a high pressure at diffuser exit are desirable in an ejector expansion refrigeration cycle. The analysis for the ejector performance with motive and suction nozzle efficiencies of 0.9, 0.8 and 0.7, mixing section efficiency of 1 and ejection ratio of 0.3 was repeated.

Based on the modeling results, it is found that higher isentropic nozzle efficiencies are desirable in an ejector expansion refrigeration cycle as shown in Figures 2 to 4. Figure 2 and 3 shows that the optimum values of the motive nozzle throat diameter and the mixing section diameter varies around 2.4 mm and 3.5 mm at different nozzle efficiencies, respectively. It can be seen from Figure 4 that diffuser exit pressure increases quickly and then slowly, while the diffuser exit quality decreases quickly and then slowly with the increase of the diffuser area ratio.

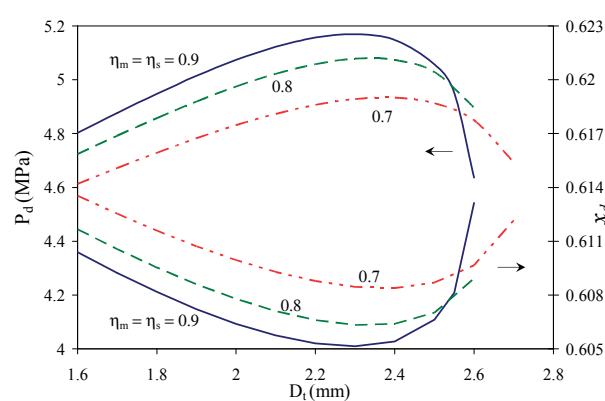


Figure 2: Ejector discharge pressure and quality versus motive nozzle exit diameter
($P_i = 9.5$ MPa, $T_i = 313.15$ K, $P_s = 3.969$ MPa, $\phi = 0.3$,
 $D_m = 4$ mm, $D_d = 8$ mm)

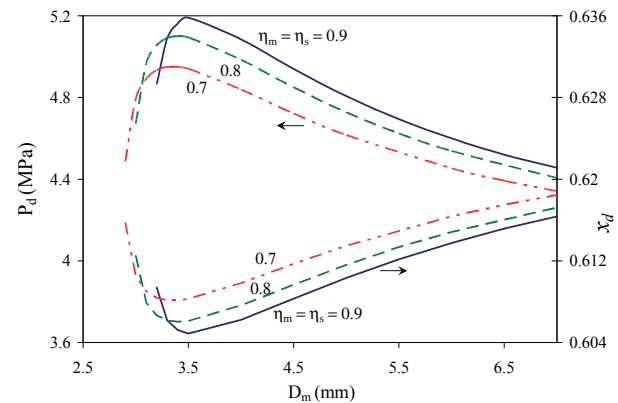


Figure 3: Ejector discharge pressure and quality versus mixing section constant area diameter
($P_i = 9.5$ MPa, $T_i = 313.15$ K, $P_s = 3.969$ MPa, $\phi = 0.3$,
 $D_t = 2$ mm, $D_d = 16$ mm)

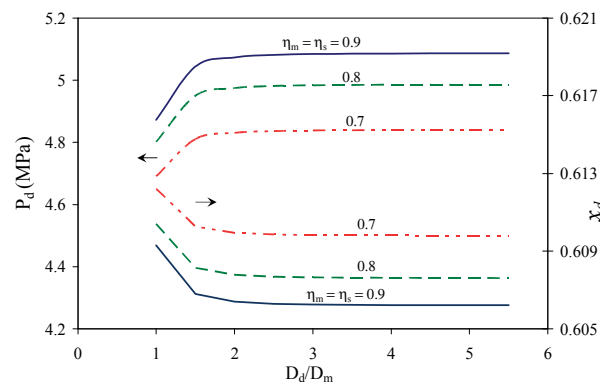


Figure 4: Ejector discharge pressure and quality versus diffuser diameter ratio
($P_i = 9.5$ MPa, $T_i = 313.15$ K, $P_s = 3.969$ MPa, $\phi = 0.3$, $D_t = 2$ mm, $D_m = 4$ mm)

3. CO₂ EJECTOR EXPANSION EXPERIMENTAL SETUP

To obtain values for the ejector motive and suction nozzle efficiencies and the mixing section efficiency, a controllable ejector expansion device was designed, constructed, and installed in an existing transcritical CO₂ bread board Environmental Control Unit (Li and Groll 2004). In addition, a new separator was installed at the outlet of the ejector. Afterwards, the ejector-expansion ECU was tested at various operating conditions.

3.1. Description of the Controllable Ejector Expansion Device

The stainless-steel controllable ejector expansion device constructed by the Mechanical Engineering machine shop at Purdue University is presented in Figure 5. Detailed drawings of the ejector expansion device, including drawings of the motive nozzle, suction nozzle-mixing section-diffuser, needle, and connectors, are presented in Liu and Groll (2008). The ejector expansion device was installed in the transcritical CO₂ bread board ECU using Swagelok NPT thread connectors.



Figure 5: Photograph of controllable ejector expansion device

3.2. Description of the ejector expansion CO₂ ECU test setup

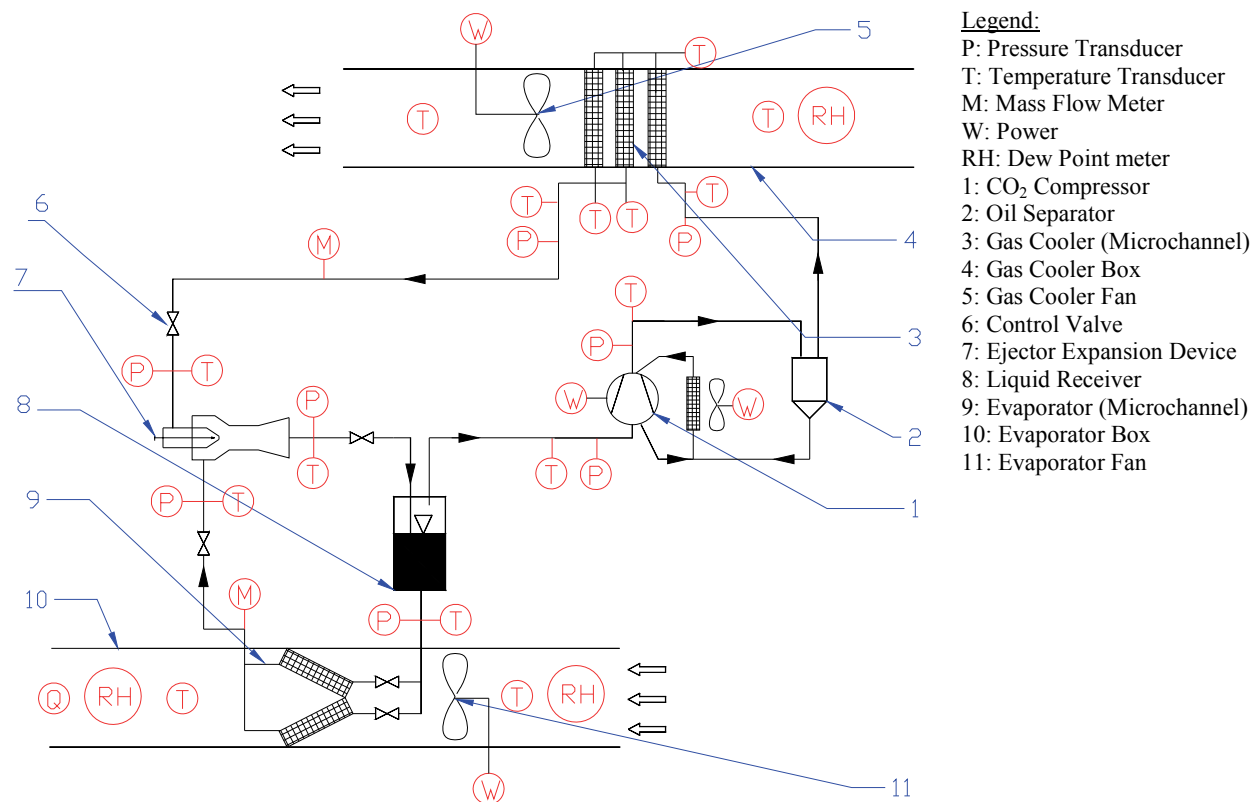


Figure 6: Schematic of ejector expansion CO₂ ECU test setup

The transcritical CO₂ bread board ECU consists of an indoor unit and an outdoor unit, which are located in the two side-by-side psychrometric chambers. The indoor unit consists of an evaporator, evaporator box, evaporator fan, expansion valve, bypass valve and liquid receiver. The outdoor unit consists of a gas cooler, gas cooler box, gas cooler fan, and compressor and oil separator. A detailed description can be found in Li and Groll (2004). A schematic of the ejector expansion CO₂ ECU test setup is shown in Figure 6, where the expansion valve was replaced by the controllable ejector expansion device.

3.3. Ejector Test Data Reduction

The test data recorded during the two-phase flow ejector tests are the CO₂ pressures and temperatures, and the mass flow rates at the inlets to the motive nozzle and the suction nozzle as well as the CO₂ pressure at the ejector outlet. The two-phase flow ejector model was used to determine the motive nozzle, suction nozzle and mixing section efficiencies based on the measured data. The overall flow chart to determine the internal ejector efficiencies is shown in Figure 7. The isentropic efficiency of the motive nozzle was determined by matching the measured motive nozzle mass flow rate to the motive nozzle mass flow rate predicted using the two-phase flow ejector model. The isentropic efficiency of the suction nozzle was determined by matching the measured suction nozzle mass flow rate to the suction nozzle mass flow rate predicted using the two-phase flow ejector model. The mixing section efficiency was determined by matching the measured ejector outlet pressure to the predicted ejector outlet pressure using the two-phase flow ejector model.

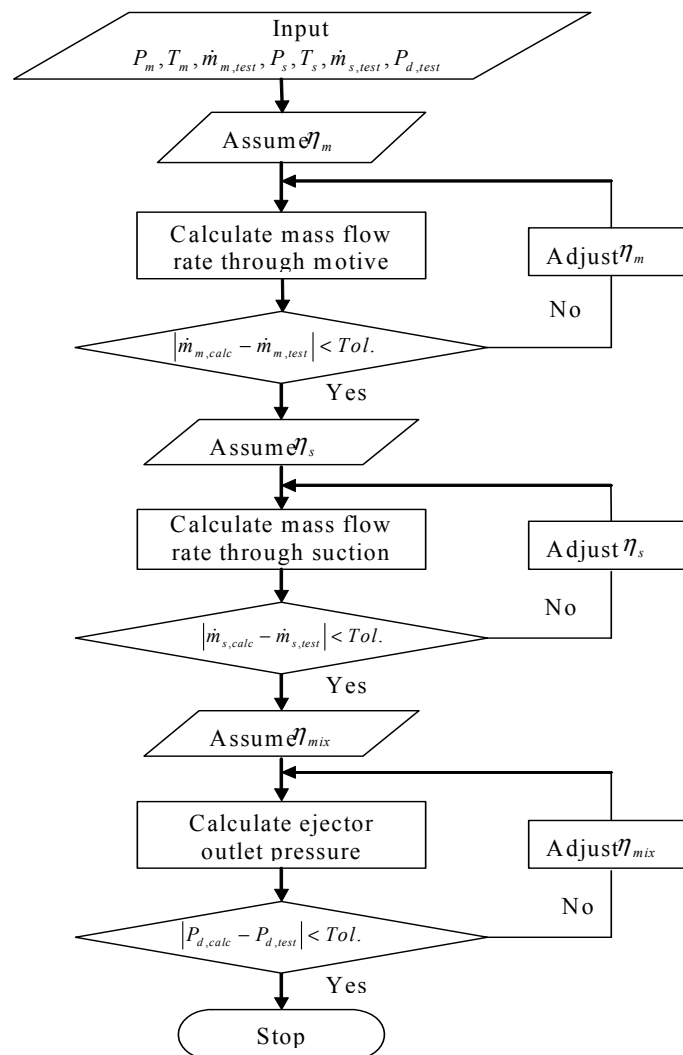


Figure 7: Flow chart to determine motive and suction nozzle isentropic efficiencies as well as mixing section efficiency

3.4. Uncertainties of Ejector Efficiencies

Table 1 presents the calculated ejector nozzle and mixing section efficiencies based on the measured values of one of the test runs and the uncertainties that are associated with these efficiencies based on the uncertainties of the individual measurements. The uncertainties of the efficiencies were determined using a standard error analysis in EES (Klein 2004). It can be seen from Table 1 that the nozzle efficiencies and mixing section efficiency can be determined within $\pm 6\%$ given the listed accuracy of the various measurement instrumentations. It can also be seen that the uncertainties associated with the motive nozzle inlet temperature of $\pm 0.5^\circ\text{C}$, L2 of $\pm 0.2\text{ mm}$, and motive nozzle and suction nozzle inlet pressures of $\pm 0.019\text{ MPa}$ are the most significant contributions to the final uncertainties of the calculated motive nozzle isentropic efficiency η_m , motive nozzle isentropic efficiency η_s and mixing section efficiency η_{mix} , respectively. L2 is the distance from the motive nozzle exit to the mixing section inlet as shown in Figure 8.

Table 1: Uncertainty analysis of ejector components

Measured data	Value	Absolute uncertainty	η_m	η_s	η_{mix}
			Uncertainty Contributions		
L1 (mm)	54.03	0.2	0.01481%	0.00%	0.01472%
L2 (mm)	38.3	0.2	0.00%	97.64%	0.00%
P _m (MPa)	12.855	0.019	17.72%	0.00%	47.16%
P _s (MPa)	3.748	0.019	0.00%	0.8649%	47.15%
T _m (°C)	50.88	0.5	69.33%	0.00%	3.279%
T _s (°C)	21.63	0.5	0.00%	0.05455%	0.00%
\dot{m}_m (kg/s)	0.18	0.0008	12.94%	0.00%	0.1062%
\dot{m}_s (kg/s)	0.07	0.00035	0.00%	1.444%	1.607%
P _o (MPa)	4.499	0.019	0.00%	0.00%	0.6791%
Calculated Results			0.986	0.972	0.882
Absolute Uncertainty			0.01056	0.05734	0.02696
Relative Uncertainty			1.071%	5.9%	3.058%

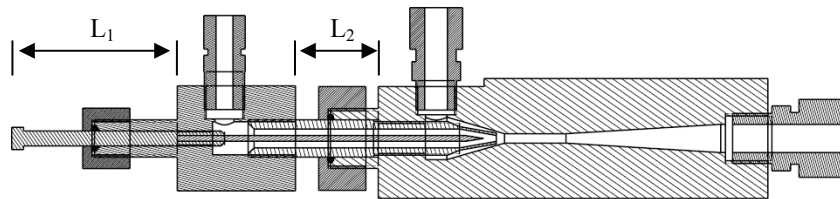


Figure 8: Schematic of controllable ejector expansion device

4. RESULTS FOR EJECTOR EFFICIENCIES

Figure 9 shows the motive nozzle and suction nozzle efficiencies as well as mixing section efficiency determined by the measured parameters for 29 test runs conducted with the transcritical CO₂ bread board ECU. The efficiencies of the ejector obtained at different outdoor temperatures are indicated in Figure 9 by using one symbol for each outdoor temperature. It can be seen that the motive nozzle efficiency decreases as the ejector throat diameter decreases, which reduces the pressure ratio of the motive nozzle inlet pressure to suction nozzle inlet pressure increase. The suction nozzle efficiency is affected by the suction nozzle inlet pressure (related to indoor temperature), ejector throat area, and motive nozzle exit position relative to the mixing section constant area entry. The mixing section efficiencies at a motive nozzle exit distance from the mixing section constant area entry of 1.5 times of the mixing section constant area diameter are generally higher than those at a distance of 6 times of the mixing section constant area diameter. The ejection ratio reaches its highest value at the smallest throat diameter.

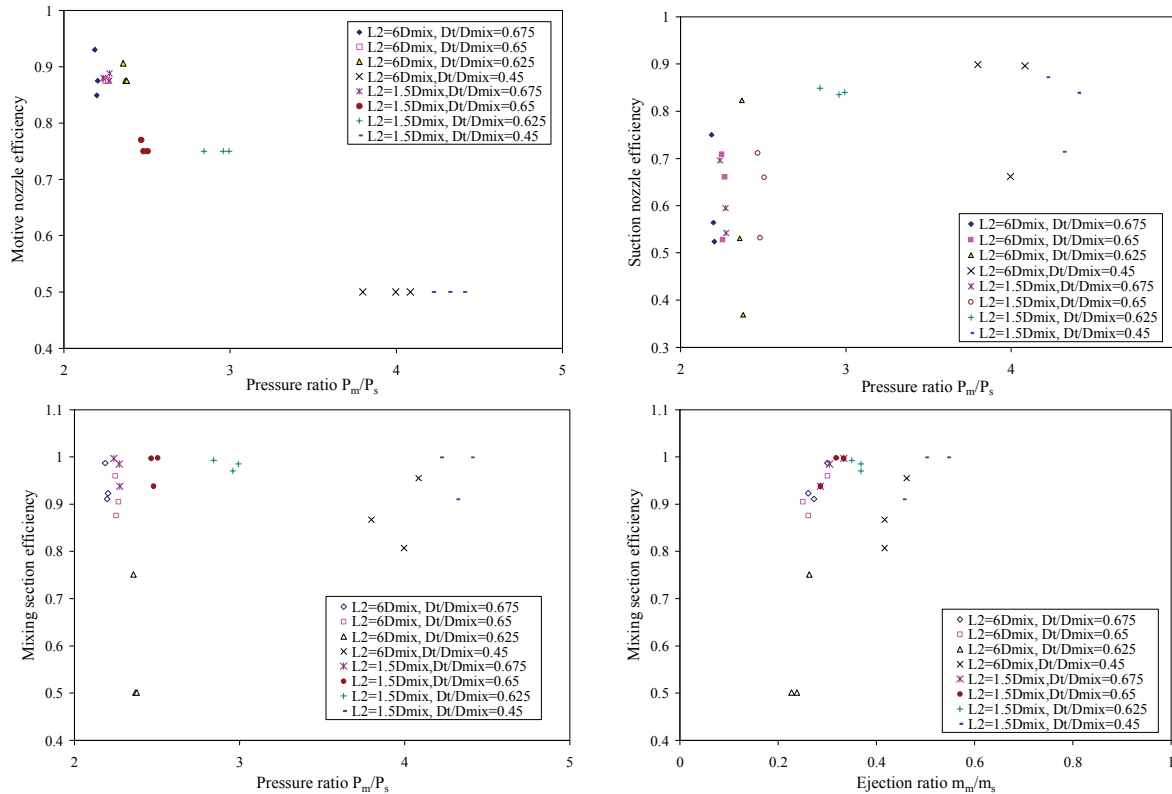


Figure 9: Ejector efficiencies of motive nozzle, suction nozzle and mixing section determined from test data

5. CONCLUSIONS

A two phase flow ejector model for the transcritical CO_2 cycle was developed. The effects of ejector geometries on CO_2 ejector performance were studied. The optimum values of the motive nozzle throat diameter and the mixing section diameter occur approximately at 2.4 mm and 3.5 mm, respectively. The diffuser exit pressure increases quickly and then slowly as a function of the diffuser diameter ratio D_d/D_m . A controllable ejector expansion device was designed, constructed and incorporated into an experimental CO_2 ECU setup. The ejector efficiencies were determined using the measured data. The pressure ratio of the motive nozzle inlet pressure to suction nozzle inlet pressure increases as the motive nozzle throat diameter decreases, which results in a decrease of the motive nozzle efficiency. The suction nozzle efficiency is affected by the suction nozzle inlet pressure and ejector throat area. The ejector with the motive nozzle exit distance from the constant area mixing section entry of 1.5 times of the mixing section constant area diameter has a higher mixing section efficiency than the ejector with the distance of 6 times of the mixing section constant area diameter.

NOMENCLATURE

COP	coefficient of performance	D	Diameter (mm)
h	specific enthalpy (Kj/kg)	\dot{m}	mass flow rate (kg/s)
P	Pressure (MPa)	T	temperature ($^{\circ}\text{C}$)
v	specific volume	χ	quality

Subscripts

b	suction nozzle exit	c	critical	d	diffuser
dis	discharge	f	fluid	g	vapor
m	motive nozzle	mix	mixing section	i	inlet
is	isentropic	s	suction nozzle	t	nozzle throat

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